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DEVELOPMENT OF A LABORATORY TEST TO PREDICT SCUFFING PERFORMANCE OF HIGH-TEMPERATURE ENGINE OILS

INTERIM REPORT
TFLRF No. 322

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Under Contract to
U.S. Army TARDEC
Mobility Technology Center–Belvoir
Fort Belvoir, Virginia

Contract No. DAAK70-92-C-0059

January 1997

Disclaimers

DTIC Availability Notice

Disposition Instructions

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (Leave blank)

2. REPORT DATE

January 1997

3. REPORT TYPE AND DATES COVERED

Interim
July 1993 to July 1996

4. TITLE AND SUBTITLE

Development of a Laboratory Test to Predict Scuffing Performance of High-Temperature Engine Oils

5. FUNDING NUMBERS

DAAK70-92-C-0059; WD 19

6. AUTHOR(S)

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7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

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8. PERFORMING ORGANIZATION REPORT NUMBER

TFLRF No. 322

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

Department of the Army
Mobility Technology Center-Belvoir
10115 Gridley Road, Suite 128
Ft. Belvoir, Virginia 22060-5843

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Approved for public release; distribution unlimited

12b. DISTRIBUTION CODE

13. ABSTRACT (Maximum 200 words)

The requirements placed on lubricating oils at the top ring reversal point are becoming increasingly severe due to recent changes in engine technology, most of which are driven by emissions regulations. As a result, revised engine tests to evaluate cylinder liner scuffing are being introduced to engine oil specifications. Despite the high cost and generally poor repeatability of full-scale engine data, no laboratory screener test is commercially available. In the present study, various lubricant characteristics were correlated with the level of scuffing in the Detroit Diesel 6V92TA engine test. Good initial agreement was obtained with lubricant volatility measured at 525°C, which is predicted to be the approximate contact temperature under the most extreme conditions likely to exist during normal operation.

Measurement of lubricant volatility, however, yields no indication of additive response and also appears to become less accurate when applied to unconventional basestocks. As a result, a laboratory-scale wear test was developed to predict scuffing resistance under high stress conditions. The results of wear and volatility tests are combined using a simple equation to form the Diesel Engine Oil Scuff Test (DEOST). The resulting methodology provides an R^2 correlation of 70 percent with scuffing measured in the 6V92TA engine and is sensitive to both basestock characteristics and antiwear additives. In addition, the DEOST results indicated that viscosity index improvers provide little benefit under high-temperature operation, an effect commonly observed by failure of petroleum-based multigrade oils in the 6V92TA engine test.

14. SUBJECT TERMS

Wear	Piston-Ring	Lubrication
Friction	Volatility	Engine
Scuffing		

15. NUMBER OF PAGES

41

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	

EXECUTIVE SUMMARY

Problem: Adiabatic and partially cooled compression ignition engines are currently being designed for use in military vehicles. The increased operating temperatures produced by this technology requires new lubricant additive packages and basestock formulations. A number of full-scale engine tests are available to evaluate lubricant performance under both conventional and severe operating conditions. However, the relatively high cost and poor repeatability of these procedures makes them unsuitable for screening new and unproven lubricants.

Objective: The objective of this project is to develop a laboratory-scale test procedure capable of predicting scuffing and adhesive wear resistance provided by lubricant test candidates. The test methodology must be both cost-effective and repeatable, while also providing good correlation with scuffing observed in full-scale engines.

Importance of Project: Current engine design is limited by lubrication technology, particularly at the critical contact between the piston-ring and cylinder liner. Liquid lubricants remain the only practical means of ensuring both low friction and wear at this contact. Development of new lubricant formulations would be greatly assisted by a cost-effective laboratory-scale lubricant screening test.

Technical Approach: The failure mechanisms present in full-scale engine tests were defined through observation and simple mathematical modelling. The Diesel Engine Oil Scuff Test (DEOST) was then developed, cognizant of the lubrication mechanisms present at the piston-ring and cylinder liner contact. The accuracy of the DEOST methodology was validated using oils of known quality, as defined using the 6V92TA engine test.

Accomplishments: Good agreement was demonstrated between the DEOST procedure and the 6V92TA engine test, with an R^2 correlation of 70 percent. This correlation is partially degraded by the repeatability errors present in the full-scale engine data. The DEOST procedure is sensitive to both lubricant volatility and wear resistance, which were shown to be the primary factors affecting scuffing in the engine. A number of recently developed advanced high-temperature lubricants intended for use in partially cooled military engines were also evaluated. The DEOST procedure predicted that each experimental lubricant would provide excellent performance in the 6V92TA engine test.

Military Impact: The DEOST procedure will greatly simplify screening and development of new thermally stable lubricant formulations. These lubricants will allow the benefits and payoffs of minimum-cooled diesel engines to be realized. In addition, the repeatability of the DEOST procedure is measurably better than that available in the more complex full-scale engine tests. This accuracy will allow the effects of minor variations in lubricant chemistry to be easily observed.

FOREWORD/ACKNOWLEDGEMENTS

This work was performed by the U.S. Army TARDEC Fuels and Lubricants Research Facility (TFLRF) located at Southwest Research Institute (SwRI), San Antonio, Texas, during the period July 1993 through July 1996. The work was partially funded under Contract No. DAAK70-92-C-0059 by the U.S. Army TARDEC, Mobility Technology Center-Belvoir (MTCB), Fort Belvoir, Virginia. Mr. T.C. Bowen (AMSTA-RBFF) of MTCB served as the contracting officer's representative, and Mr. M.E. LePera (AMSTA-RBF) of MTCB served as the project technical monitor. Cooperative funding for this effort was provided by the Propulsion System Division, U.S. Army TACOM, with Mr. Ernest Schwarz (AMSTA-TR-R) serving as the TACOM project technical monitor.

The author would like to acknowledge the efforts of TFLRF and SwRI personnel, including Mr. R.T. Stockwell, who provided technical support; Mr. T.E. Loyd, who performed the bench wear tests; Ms. A. Ayala and Mr. J.H. Marshall, who provided administrative assistance; and Ms. M.M. Clark, who edited the final draft of the report.

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I. INTRODUCTION

In its most general sense, scuffing is considered to result from failure of the boundary lubricating film following thermal feedback: at a critical flash temperature, the surface-active lubricant additives are stripped from the surface, resulting in a further increase in friction and contact temperature and ultimately, severe adhesive wear. This mechanism most often occurs with new piston ring and liner pairs, particularly if the running-in process is not sufficiently rapid to remove non-conforming surface asperities. The traditional scuffing failure mechanism may be further aggravated by lubricant starvation. Indeed, it has been recognized since the 1950's that lubricant volatility is an important factor in both oil consumption and wear resistance, related through viscosity (1)* and distillation (2, 3).

Newer engines designed to minimize particulate emissions typically have retarded injection timing and position the top rings closer to the piston crown to decrease dead volume in the combustion chamber. In addition, the clearance between the piston ring and cylinder liner is reduced to minimize oil flow to this area. These changes increase the stress on the lubricant by placing a reduced oil volume further into the high-temperature zone for a longer residence time. As a result, full-scale engine tests such as that for the 6V92TA have been developed to reflect the oil's ability to protect against scuffing and scoring under high power and high load conditions.(4) The 6V92TA engine oil test is being introduced to specifications and classifications of engine lubricating oils such as ASTM D 4485 (5), SAE classification J183 (6), and Military Specification MIL-L-2104 (7).

Lubricant volatility is primarily derived from basestock characteristics, and the range of basestocks being brought to market is increasing. This change is evident from API 1509 Document (8), which has been modified to include five base oil groups from the original three. However, in practice, most crankcase lubricants are classified by viscosity grade, which is only directionally related to volatility. The issue is complicated by the fact that polymers are commonly added to light petroleum basestocks to improve viscosity index but have little effect

* Underscored numbers in parentheses refer to the list of references at the end of this report.

on volatility, while many synthetic basestocks require little or no polymeric additives to produce the viscosity index of the equivalent petroleum-based oil.(9, 10) The volatility characteristics of hydrocarbon basestocks are also affected by both crude source (i.e., the proportion of paraffinics, aromatics and polars) and processing method (i.e., the level of solvent extraction, hydrotreating, hydrocracking, hydrofinishing, and dewaxing).(11)

Phosphorous-based additives are commonly used to enhance basestock performance and significantly improve friction and wear resistance under boundary lubricated conditions. Recently, however, ILSAC GF-2 requirements for commercially available passenger car motor oils lowered phosphorous limits from a maximum of 0.12 to 0.10 percent, measurably reducing wear resistance in cam followers and to a lesser extent, at the piston ring and liner.(12) In a separate but related area, high-temperature liquid lubricants are being developed for adiabatic and partially cooled compression ignition engines intended for military applications.(13) These fluids must be capable of operating with a top ring reversal temperature of 285°C and above, which is appreciably higher than the 180 to 200°C present in most commercial engines.(14) Again, optimum thermal deposit resistance is commonly achieved in oils that contain few metal-based antiwear additives.

Clearly, a laboratory-scale Diesel Engine Oil Scuff Test (DEOST) procedure that reflects basestock characteristics as well as additive effectiveness would greatly simplify screening of effective lubricant formulations. A large number of test procedures are available to measure antiwear additive quality, particularly under steady-state conditions with low specific wear rates (i.e., in the range 10^{-10} to 10^{-13} $mm^3mm^{-1}N^{-1}$). However, a search of the literature failed to reveal many viable volatility-related bench tests that predict scuffing in an engine such as the 6V92TA.(3) The objective of the present study is to develop such a test for use in both military and commercial applications.

II. EXPERIMENTAL

A. Test Lubricants

A total of 25 commercially available crankcase lubricants of widely varying quality were procured by the U.S Army. The results of preliminary tests to define the basic characteristics of each lubricant are provided in TABLE 1. Kinematic viscosity was defined at 40 and 100°C according to ASTM D 445 (15), while the high shear viscosity was defined using ASTM D 4624 (16). The matrix of test oils incorporates a wide range of characteristics and includes low viscosity fluids intended for use in Arctic conditions, multigrade oils, as well as higher viscosity monograde oils. The boiling point distribution of each oil was defined using gas chromatography (GC), following the procedure defined in ASTM D 2887 (17) but modified to allow evaluation of fluids with a final boiling point of up to 700°C.

B. Engine Tests

Each of the 25 oils in TABLE 1 had been evaluated using either the 6V92TA or 6V53T engine test procedures summarized in TABLE 2. Both methods define the high-temperature performance characteristics of compression ignition engine lubricants, particularly cylinder liner scuffing and piston ring face distress. The reported result reflects the fraction of the cylinder liner damaged by scuffing relative to the total circumference of the liner. Four oils had been evaluated for use in Arctic conditions by the U.S. Army using the 6V53T engine test. The remaining 21 oils were evaluated using the 6V92TA engine, which is required for qualification of commercial engine oils.(4) This procedure has a 95 percent confidence interval of ± 18 percent and a pass/fail cut-off limit of 45 percent. The 6V92TA procedure was derived from the 6V53T engine test, but uses a simplified procedure to rate cylinder liner distress. Perfect correlation does not exist between the two engines; however, directional agreement has been demonstrated.(18) To simplify data analysis, the results for the 6V53T engine tests were adjusted to be comparable to those of the 6V92TA.

TABLE 1. Laboratory Test Results

Code	SAE Grade	6V92T, %	LFW-1 Results, mm Run A	Run B	Average	Four-Ball, mm	Kin. Visc., cSt, °C 40	100	Visc. Index	HSV*	Boiling Point Dist., °C IBP†	50	80	100
DD 6V92TA Engine Test Data														
C-1	40	8	1.02	0.71	0.86	0.35	149	14.8	99	4.58	305	510	546	599
C-2	30	20	0.88	1.00	0.94	0.40	106	12.2	104	3.70	267	494	539	707
C-3	40	15	1.25	0.75	1.00	0.43	145	14.7	100	4.38	318	503	539	609
C-4	30	37	1.14	0.95	1.04	0.44	90	10.8	105	3.36	309	496	539	603
C-5	15W-40	34	1.32	1.29	1.31	0.38	112	14.8	136	4.35	264	439	497	710
C-6	30	25	1.51	1.30	1.40	0.45	87	11.3	118	3.67	299	421	538	612
C-7	30	45	1.51	1.40	1.45	0.39	101	11.6	102	3.57	291	465	497	597
C-8	30	27	1.56	1.40	1.48	0.46	97	11.5	106	3.66	321	479	523	598
C-9	15W-40	43	1.68	1.30	1.49	0.45	105	14.1	136	4.05	317	437	469	570
C-10	15W-40	47	1.41	1.70	1.55	0.46	116	15.2	136	4.09	283	422	461	606
C-11	15W-40	60	1.61	1.50	1.55	0.43	112	14.6	133	4.41	320	451	497	706
C-12	15W-40	49	1.72	1.60	1.66	0.46	111	14.7	136	4.12	287	427	465	598
C-13	30	26	1.7	1.67	1.68	0.46	96	11.0	100	3.40	331	478	520	601
C-14	15W-40	63	1.75	1.66	1.68	0.43	--	--	--	--	277	457	502	722
C-15	15W-40	45	1.61	1.80	1.70	0.44	114	15.4	141	4.14	305	428	466	605
C-16	30	30	1.83	1.59	1.71	0.44	108	12.1	102	3.73	300	481	535	611
C-17	15W-40	43	1.82	1.67	1.74	0.44	--	--	--	--	335	474	516	721
C-18	15W-40	35	1.68	1.91	1.79	0.46	117	15.7	141	4.41	308	427	463	602
C-19	15W-40	63	1.85	1.80	1.82	0.46	117	15.6	140	4.1	295	431	470	582
C-20	15W-30	60	1.99	1.78	1.88	0.46	101	13.9	141	4.06	314	423	442	557
C-21	15W-40	58	2.01	2.15	2.08	0.43	97.4	13.8	143	3.64	295	410	456	609
DD 6V53T Engine Test Data														
C-22	OEA-30	30‡	1.04	1.09	1.07	0.38	53	10.1	180	3.22	302	466	487	594
C-23	0W-30	60‡	1.62	1.57	1.59	0.47	51	11.0	215	3.01	273	418	431	597
C-24	10	63‡	1.75	1.60	1.67	0.45	42	6.95	189	2.45	306	413	444	567
C-25	OEA-30	30‡	1.76	1.08	1.42	0.37	51	10.1	189	3.21	327	461	487	703

* High-Shear Viscosity
† Initial Boiling Point
‡ Data points estimated from results obtained in DD 6V53T engine test.

TABLE 2. **Summary of Engine Conditions Used to Evaluate Test Lubricants**

Parameter	6V92TA	6V53T	SCE 903
Sump Temp., °C	125	121 (max.)	171
Approx. TRR Temp., °C	180 to 200	180 to 200	250
Horsepower	500	300	167
Cylinders	6	6	1
Bore, mm	123	90	139
Stroke, mm	127	114	121
Duration, hr	100	240	200
Modes, hr @ RPM	Intermittent	Intermittent	Continuous
Break-in	6	2	--
Idle	--	0.5 @ 1,000	--
Max. Power	8 @ 2,300	2 @ 2,800	--
Idle	--	0.5 @ 1,000	--
Max. Torque	8 @ 1,200	2 @ 2,200	200 @ 2,300
Soak	3 @ 0	4 @ 0	--

Advanced lubricants are currently being evaluated by the U.S. Army for use in future adiabatic and partially cooled compression ignition engines. As part of that study, the high-temperature performance of six oils was evaluated using a Cummins SCE 903 engine.(3) The block and cylinder head of the SCE 903 were modified to reduce heat loss and simulate high-temperature engine operation. These changes resulted in a top ring reversal temperature of 250°C and a sump temperature of 171°C, which is appreciably higher than either the 6V92TA or 6V53T engines. Although no direct comparison is available, the SCE 903 engine test would be expected to be a more severe test of lubricant performance than the 6V92TA. Each of the experimental lubricants produced little measurable scuffing in the SCE 903 engine and so would be expected to give relatively good performance in the 6V92TA. More details of the SCE 903 engine test procedure may be obtained in Reference 13.

C. Laboratory-Scale Wear Tests

The lubricity of the 25 oils was evaluated using both a four-ball and an LFW-1 wear test apparatus. The four-ball wear tests were performed according to the procedures defined in ASTM D 4172 (19) and ASTM D 2783 (20), which define the wear resistance and load carrying

capacity of the test fluid, respectively. Detailed descriptions of the four-ball test apparatus are available from a number of sources and are therefore not repeated here. The LFW-1 apparatus, shown schematically in Fig. 1, is similar to that described in ASTM D 2714 (21). A test ring is rotated against a steel block, the specimen assembly being partially immersed in the lubricant sample. When the test ring is moved against the block, the resulting friction force is measured using a load cell with output to a strip chart recorder. A ring specimen is mounted on the protruding mandrel end of a shaft supported by precision roller bearings. An opposing test block is forced against the ring by a self-aligning holder. The lubricant temperature may be adjusted from ambient to 100°C, although frictional heating commonly results in appreciably higher temperatures during testing. Spindle speed in the LFW-1 apparatus may be adjusted up to 1,200 RPM, which corresponds to a sliding speed of 2.20 m/s. Load is applied through a 30:1 compound lever system, resulting in a contact force of up to 270 kg at an applied load of 9 kg. Following each test, the average width of the wear scar on the stationary block is assessed using an optical microscope and taken as a measure of lubricant quality.

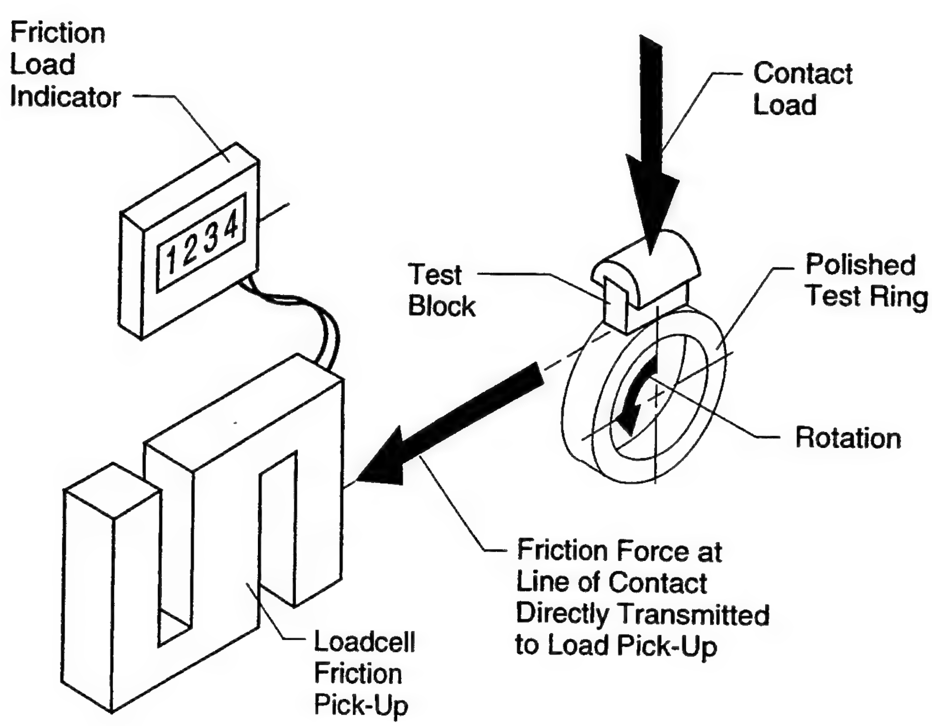


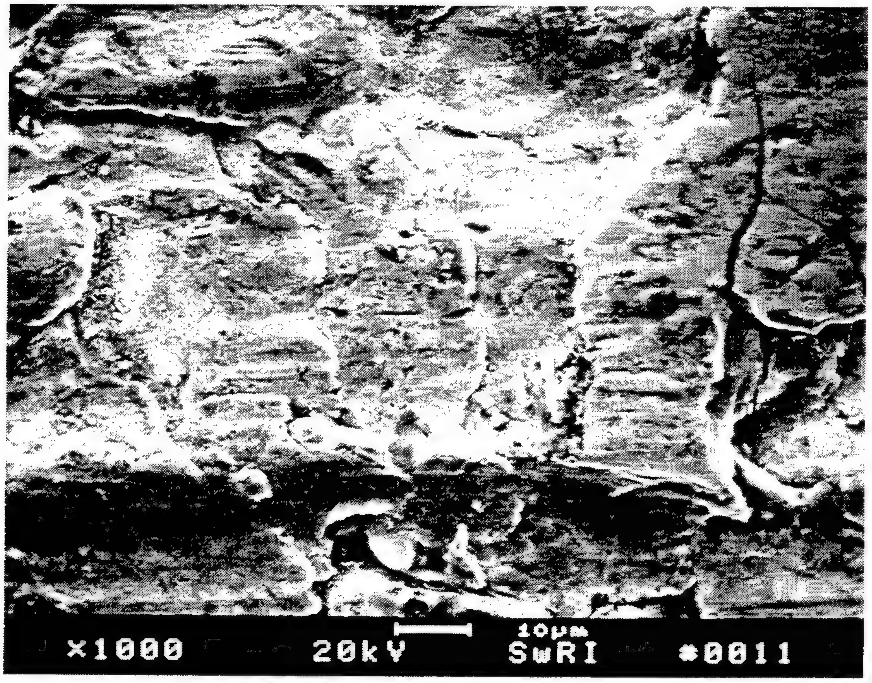
Figure 1. **Schematic diagram of the LFW-1 wear test apparatus used in the DEOST procedure**

III. TEST DEVELOPMENT

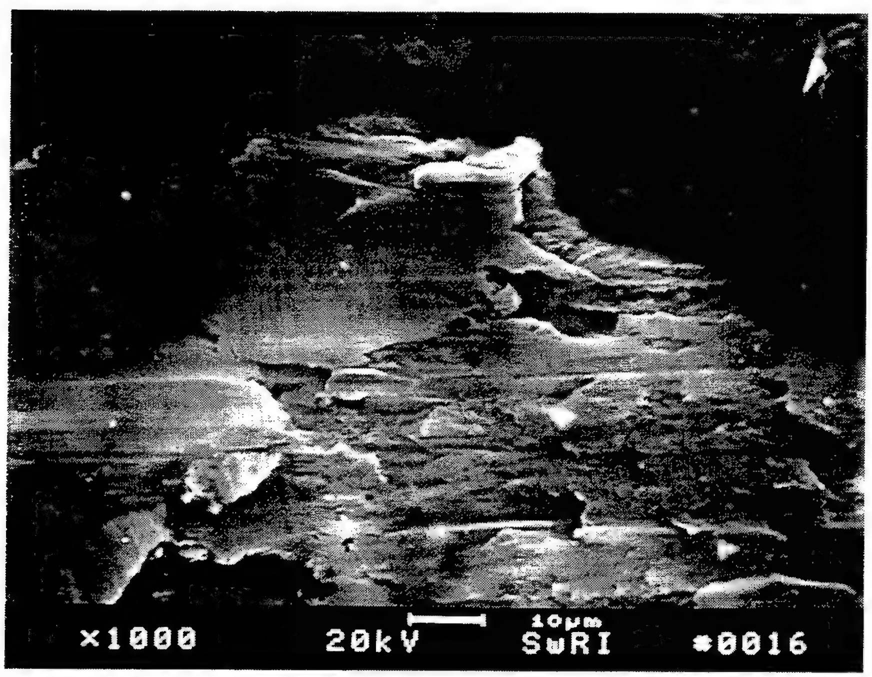
A. Wear Mechanism Evaluation in the 6V92TA Engine

Development of an effective laboratory screening test requires accurate reproduction of the wear mechanisms present in the full-scale application. Post-test cylinder liner samples from 6V92TA and 6V53T engines were sectioned and the surface topography examined using optical and electron microscopy. A photomicrograph from a partially scuffed liner is presented in Fig. 2a. Plastic deformation is apparent toward the top of the photo, with severe shearing of the near surface layers due to high friction and adhesive wear. In addition, evidence of abrasive ploughing is visible over the entire stroke length, probably due to adhesive transfer of workhardened cast iron to the hard chrome-faced ring. In many instances, the abrasion is sufficient to partially mask the original adhesive wear. Overall, however, most severe adhesion occurs in localized areas just below top dead center on the thrust side of the cylinder liner. The topography of the unscuffed portion is normally smooth or polished, with discrete graphite flakes visible.

At a critical hydrodynamic film separation, asperity interaction between the piston ring and cylinder liner occurs. The commonly used λ value in Equation 1 describes the ratio between lubricant film thickness and surface roughness. For surfaces with a Gaussian height distribution, no asperity interactions are expected if λ is greater than approximately 2.5, while values less than unity indicate considerable intermetallic contact. A mathematical analysis of the hydrodynamic film thickness at the ring liner interface in the SCE 903 engine operating at 2,300 RPM is presented in Fig. 3, modelled following the procedure described in Reference 22. A significant oil film is temporarily maintained at both top and bottom dead center due to squeeze film action. Minimum film thickness is approximately 0.5 mm $\times 10^{-3}$, typical of that predicted for many engines (23). A surface roughness of 1.1 to 1.7 μm Ra is specified for the unworn cylinder liners, which corresponds to $\lambda \approx 0.35$. However, following break-in, polishing occurs and surface roughness is commonly reduced to less than 0.2 μm Ra, which corresponds to $\lambda > 2.5$. This



a. Cast iron cylinder liner from 6V92TA engine



b. Cylinder specimen from LFW-1 wear test

Figure 2. <u>Photomicrographs of worn surface topography at 25× magnification</u>

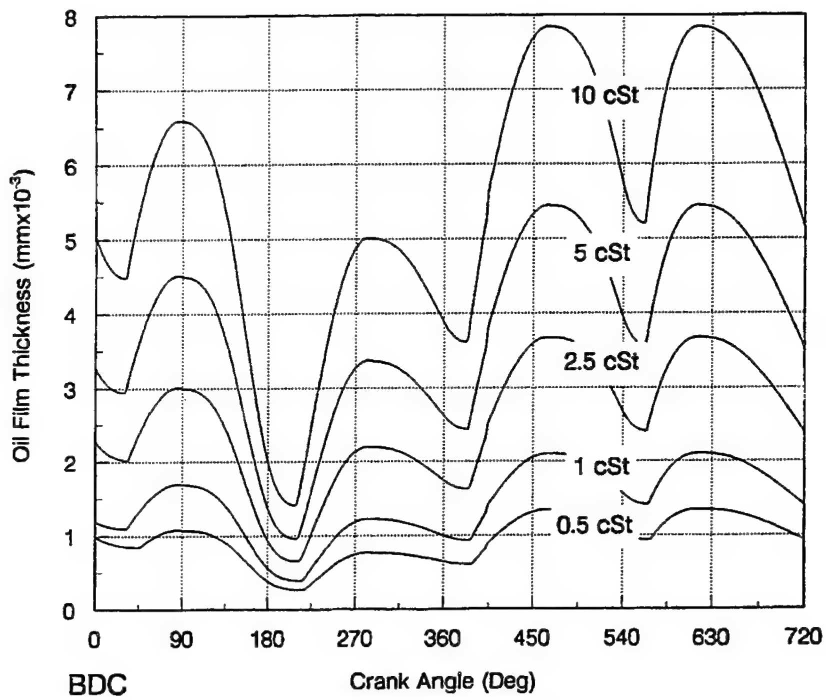


Figure 3. Theoretical oil film thickness throughout combustion cycle at 2,300 RPM

result would indicate that break-in decreases the roughness of the cylinder liner to the point where the λ ratio again reaches an acceptable value. Under these conditions, scuffing should not occur if sufficient lubricant is present.

$$\lambda = h/\sqrt{(R_1)^2 + (R_2)^2} \qquad \text{(Eq. 1)}$$

where

h = lubricant film thickness
R_1 = surface roughness of the cylinder liner
R_2 = roughness of the piston ring, which is negligible.

In an effort to minimize particulate emissions, the top ring reversal point in many engines operates under partially starved conditions and boundary lubricated sliding. The contact flash temperature at the ring liner interface may be approximated using the relationships developed by Blok, assuming a uniform pressure distribution around the ring face.(24) Maximum flash temperature is likely to occur at the point of minimum film thickness, or approximately 30° after top dead center due to the dynamics of the crank mechanism combined with increasing

combustion pressure. In the current test engines, this corresponds to a sliding speed of 10 m/s and an instantaneous combustion pressure of approximately 6.5 MPa. Blok's theory predicts a maximum ring face temperature of 356°C for a mean cylinder liner temperature of 180°C and a friction coefficient of 0.12 under these conditions.

Occasionally, however, an isolated area of increased contact stress occurs. The resulting tangential stress causes an area of increased flash temperature, which in turn further expands the asperity height to produce a "thermal bump." The surface finish no longer fits the commonly applied Gaussian height distribution, and increased wear occurs at that point.(25) In some instances, the level of distress is sufficient to produce localized welding or scuffing, resulting in the visible formation of austenite/martensite and discoloration of the liner.(26) Korovchinski proposed the dimensionless thermal bulging parameter, which may be used to develop a correction factor to account for the effects of thermoelastic distortion of the ring liner contact.(27) This theory is best suited to contacts for which the surface deformation is relatively small compared to the thickness of the object, i.e., a rigid body. However, it may be assumed that the piston ring is rigid and free of gross elastic deformation over a length of approximately 3 cm. The localized piston ring flash temperature corrected using this model is approximately 430°C for a friction coefficient of 0.12, rising to 510°C for a friction coefficient of 0.15. Indeed, the presence of a "white layer" implies that during scuffing, the contact flash temperature has exceeded the austenite transformation temperature, which is approximately 750°C.(26)

B. Correlation of Lubricant Volatility With Scuffing

Partially starved lubrication conditions combined with very high contact flash temperatures emphasize the effects of basestock volatility on wear. The R^2 correlation observed between the 6V92TA and volatility fraction measured using GC is plotted in Fig. 4 as a function of temperature. Optimum correlation was observed at a distillation temperature between 475 and 525°C. The ASTM D 4485 specification for lubricating oils intended for use in gasoline engines mandates volatility measurement at a lower temperature of 371°C. The correlations between the 6V92TA cylinder liner scuffing and boiling point fractions measured at 371 and 525°C are

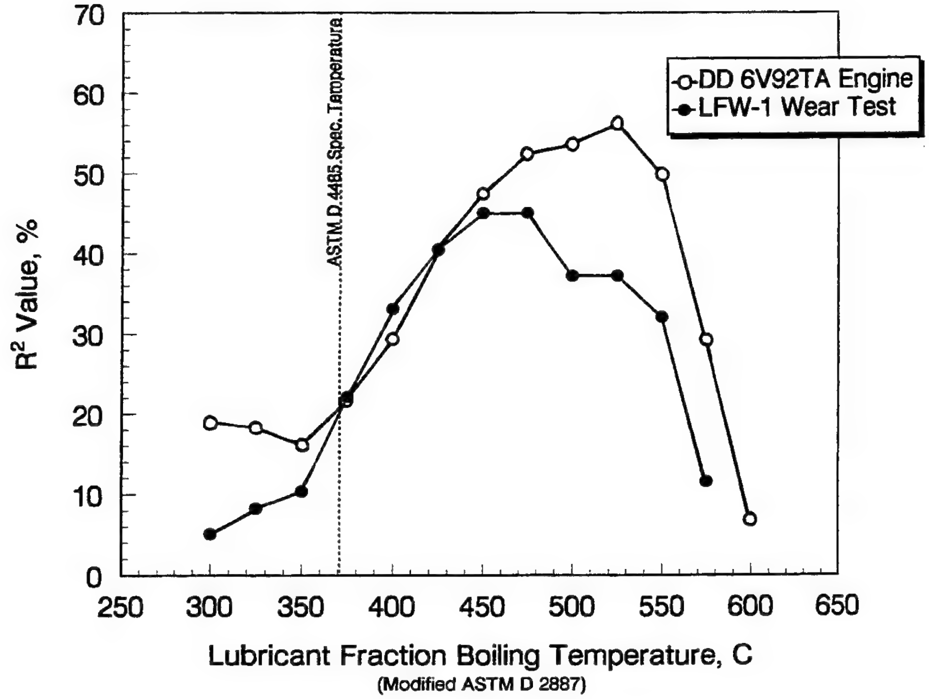
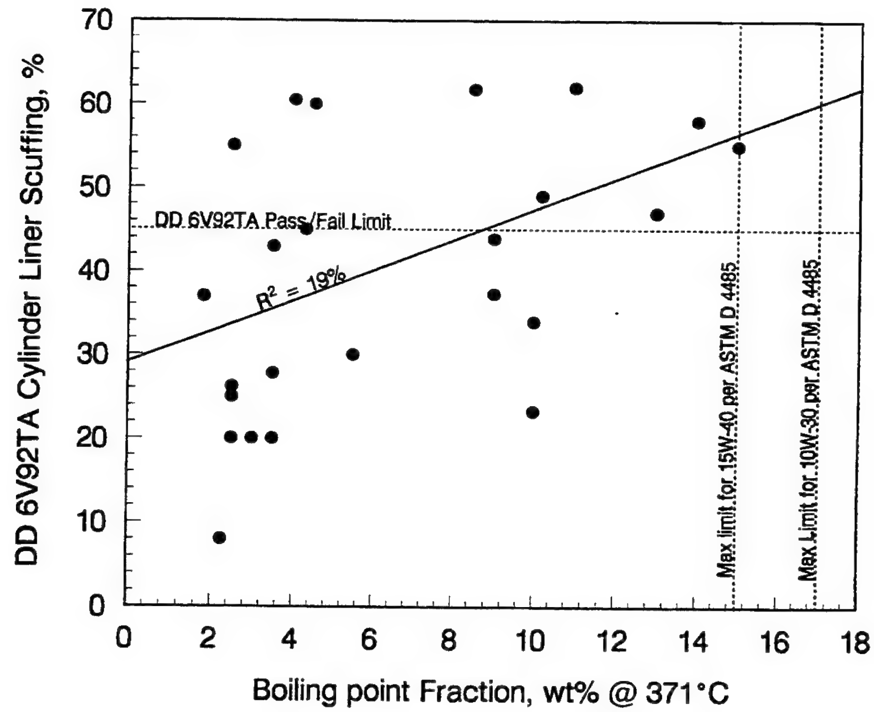


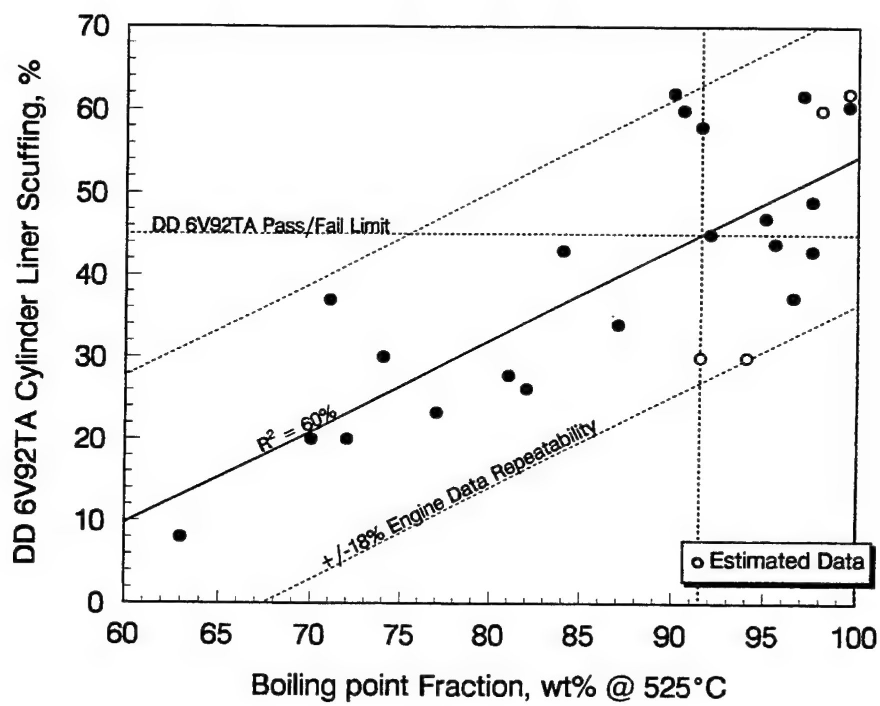
Figure 4. Correlation between cylinder liner scuffing and lubricant fraction boiling temperature measured using gas chromatography

plotted in Figs. 5a and 5b, respectively. Each of the oils in the current test matrix is well below the maximum volume-off at 371°C specified in ASTM D 4485. However, relatively poor agreement is observed between scuffing and volume-off at this temperature. In contrast, excellent correlation was observed at 525°C for all but experimental lubricants, which are not plotted. This value is considerably higher than the temperature of the bulk material at the top ring reversal point, but is very similar to the flash temperature predicted to occur at the ring/liner contact. The pass/fail limit for the 6V92TA is 45 percent scuffing, which approximately corresponds to a typical lubricant boiling point fraction of 91 percent measured at 525°C.

Relatively poor correlation was observed between most other laboratory tests and the level of scuffing observed in the 6V92TA, as shown in TABLE 3. Best agreement was observed with viscosity index, resulting in an R^2 correlation of 50 percent. This relationship reflects the widely recognized poor performance of petroleum-based multigrade oils in this engine, as may be seen from TABLE 1. Laboratory-scale wear tests were also performed using the four-ball apparatus at the contact conditions defined in TABLE 4. Some directional agreement was



a. Volume-off at 371°C



b. Volume-off at 525°C

Figure 5. Correlation between cylinder liner scuffing and lubricant boiling point fraction

TABLE 3. Correlation of Lubricant Characteristics With Scuffing

Characteristic	ASTM Method	R^2 Correlation, % 6V92TA	LFW-1*
DEOST	--	70	--
LFW-1*	--	54	--
Fraction Distilled @ 525°C	D 2887	60	46†
Fraction Distilled @ 371°C	D 2887	19	16
Four-Ball Wear Test, 40 kg	D 4172	40	40
Four-Ball EP Test	D 2783	20	27
Viscosity			
40°C	D 445	19	14
100°C	D 445	3.2	7.8
Viscosity Index	D 2270	50	22
HTHS Viscosity	D 4624	13	0.10
Initial Boiling Point	D 2887	14	0.05

* DEOST conditions from TABLE 4
† Measured at 450°C

TABLE 4. Laboratory-Scale Wear Test Conditions

Parameter	Four-Ball Wear Test	Four-Ball EP Test	LFW-1 (per ASTM)	LFW-1 (DEOST)
ASTM Procedure	D 4172	D 2783	D 2714	SwRI*
Rotational Speed				
RPM	1,200	1,760	72	600
m/s	0.46	0.65	0.131	1.1
Temperature, °C	75	18 to 35	43.3	70
Fluid Volume, mL	20	20	40	40
Duration, min	60	0.16/increment	70	20
Contact Load, kg	40	8 to 80	68	45, 60, 75
Metallurgy				
Upper Specimen	AISI E 52100	AISI E 52100	SAE 01 (H30)	SAE 01 (H60)
Lower Specimen	AISI E 52100	AISI E 52100	SAE 4620	SAE 4620
Surface Finish, µm				
Upper Specimen	0.034	0.034	0.1 to 0.2	0.1 to 0.2
Lower Specimen	0.034	0.034	0.15 to 0.30	0.04

* A more detailed description of the LFW-1 wear test procedure may be obtained in the Appendix.

produced with cylinder liner scuffing, resulting in a R^2 correlation of 40 percent. However, examination of the wear scars on the four-ball test specimens showed no evidence of adhesive wear or scuffing. Subsequent four-ball tests were performed as a function of applied load to define the extreme pressure (EP) characteristics of the test lubricants. Once again, poor correlation was observed with cylinder liner scuffing in the 6V92TA engine. Both four-ball test procedures generate relatively high contact flash temperatures and are a good reflection of the lubricant antiwear and EP chemistry. However, the relatively low temperature of the lubricant reservoir, combined with the very small contact area (and correspondingly low applied load and energy dissipation) do not emphasize the effects of lubricant volatility.

C. LFW-1 Wear Test Development

The contact flash temperature required to initiate scuffing is appreciably higher than the average temperature of cylinder liners in most commercial engines. The majority of thermal energy comes from dissipation of friction, rather than directly from the combustion process. Assuming the effects of combustion chemistry are negligible, it should be possible to simulate the onset of scuffing using a relatively simple laboratory-scale wear test. Such a test would need to consider the effects of both lubricant volatility and antiwear additives. The block-on-rotating cylinder contact geometry of the LFW-1 wear tester produces a significantly larger wear scar than the previously described four-ball wear test (up to 16 mm^2, compared to 0.282 mm^2). This larger contact area combined with correspondingly increased applied loads results in both increased specimen temperature and longer lubricant residence time within the contact junction.

Initial LFW-1 wear tests were performed using the test specimens and conditions described in ASTM D 2714. However, little or no discrimination was evident between the best and worst oils, as defined by the 6V92TA engine. Similar results were previously obtained by Stavinoha with this apparatus.(3) Subsequent increases in both sliding speed (from 0.131 to 2.2 m/s) and hardness of the block specimen (from Rc = 30 to Rc = 60) produced only a slight improvement in the correlation achieved. Examination of the contact area on the test block indicated an abrasive wear mechanism combined with only mild adhesion. This mechanism is largely due to the texture of the ground test ring, which has a surface roughness of 0.15 to 0.3 μm Ra. This

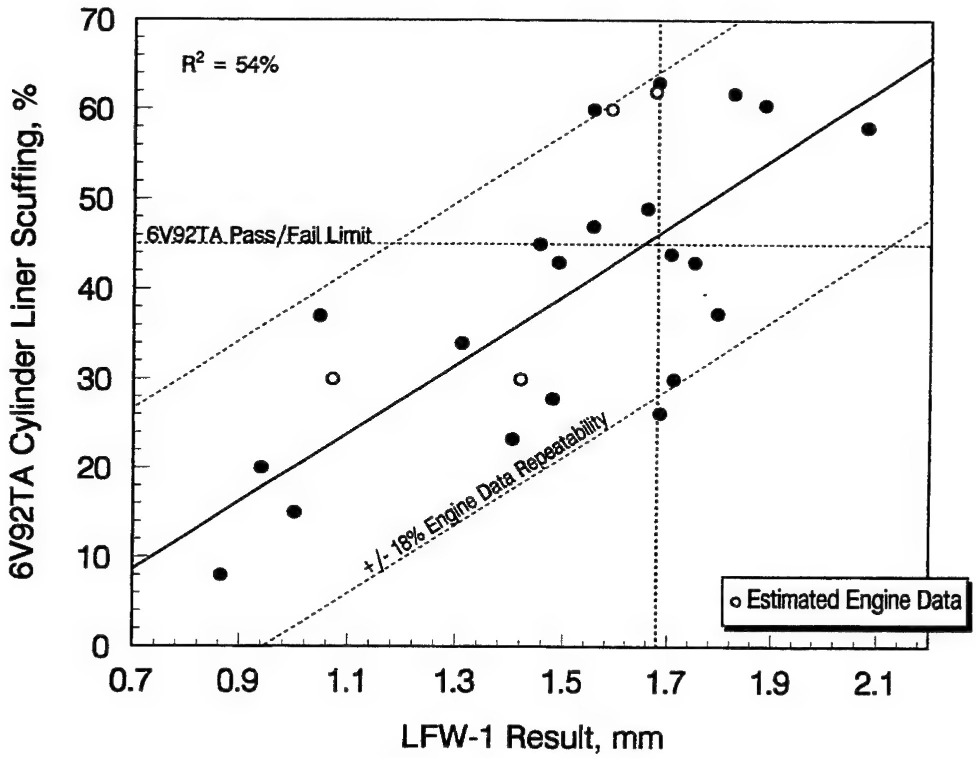
finish is rougher and more importantly, has a higher mean asperity slope than that observed on the cylinder liner following break-in. In addition, the surface texture on the test ring encourages entranement of microreservoirs of lubricant within the contact junction, thus minimizing the effects of volatility.

A large body of research has been accumulated to show that friction and wear are not simple functions of asperity height. A number of parameters have been suggested, including asperity tip slope (28) and curvature (29), average wavelength (30, 31), and correlation length (32). However, excessively smooth surfaces promote seizure due to lubricant starvation.(33) Under these conditions, elastic deformation of the surfaces allows total conformity within the contact area, thereby eliminating microreservoirs of oil between the surfaces. Consideration of these studies would indicate that the ideal surface to highlight the effects of adhesive wear and volatility would have very low surface slope while retaining some roughness. The resulting wavy surface eliminates alternate wear mechanisms and reduces the volume of lubricant within the contact junction. Such surfaces are not easily achieved through conventional processes such as grinding. However, the surface finish of many cylinder liners may approximate this ideal condition following break-in, as most short wavelength features have been removed or plastically deformed by the harder piston ring. In addition, the nature of cast iron normally results in a slightly irregular surface with depressions around graphite inclusions that trap lubricating oil.

To simulate this effect for the DEOST procedure, the surface of the LFW-1 test cylinder was first rough-ground then polished to a surface finish of approximately 0.04 μm, using 6-micron diamond paste. This methodology was previously developed for test specimens to measure scuffing load capacity.(34) The remaining parameters used in the LFW-1 wear test for the DEOST procedure are defined in TABLE 4 and include a relatively high sliding speed of 1.1 m/s at an initial lubricant temperature of 85°C. The applied load was incremented to account for the effects of decreasing contact pressure as the wear scar area increased. A more detailed description of the test methodology may be obtained in the Appendix.

D. LFW-1 Wear Test Results

The results obtained from two single LFW-1 wear tests are summarized in TABLE 1. The average value calculated from the duplicate tests is compared with the level of scuffing produced by the same oils in the 6V92TA engine test in Fig. 6 (or the estimated equivalent result for the 6V53T engine test). The best fit correlation between the laboratory and full-scale tests is represented by the solid diagonal line, which corresponds to an R^2 correlation of 54 percent. Moreover, the correlation is likely to be partially degraded by the ± 18 percent repeatability error normally associated with the engine test data. Indeed, the majority of the data points fall within the broken lines, which represent the 95 percent confidence interval for the engine test results. A scuff rating in excess of 45 percent in the 6V92TA indicates unacceptable lubricant performance. Correspondingly, an LFW-1 result greater than 1.68 mm indicates that the lubricant would, on average, fail the 6V92TA engine test.



(Note: The hollow data points represent best estimates of the 6V92TA data using results obtained from the 6V53T.)

Figure 6. Correlation between the LFW-1 wear test and cylinder liner scuffing, as measured in the 6V92TA engine test

The standard deviation of the LFW-1 procedure between the first and second runs over the 25 oils is 0.18 mm. The average of the two single test runs was used throughout the present study. This averaged result is calculated to have a standard deviation of 0.127 mm and a repeatability limit (95 percent confidence interval) of 0.356 mm. The ratio between the standard deviation and the typical span of results for the test (coefficient of variation) is 10.4 percent. This value is better than that of the engine test, which has a coefficient of variation of approximately 20 to 30 percent. Nonetheless, it is recommended that duplicate or triplicate LFW-1 wear tests be performed to establish a reliable average, thereby minimizing the effects of random variability.

Photomicrographs of the wear scar formed on the laboratory-scale wear test specimens are shown in Fig. 2b. This scar was formed using a relatively poor oil that produced high friction and wear but did not seize. Considerable plastic deformation of the AISE E 52100 steel (Rc = 60) is apparent, confirming the existence of high contact flash temperatures. The overall topography of the contact area is very similar to that shown in Fig. 2a for the scuffed area on the cylinder liner, despite the different metal used in each instance. In contrast, the more effective oils produce a smooth surface topography similar to that of a well worn-in cylinder liner, with little or no evidence of plastic deformation. Both friction coefficient and lubricant reservoir temperature are also reduced for the better oils.

The LFW-1 wear test is related to lubricant volatility, as shown in Fig. 4, resulting in an R^2 correlation of 46 percent at a lubricant fraction boiling temperature of 450°C. Blok's equation (24) predicts that the temperature of the contacting surfaces in the LFW-1 wear test may reach 500°C, for a bulk specimen/lubricant temperature of 120°C. This value assumes a friction coefficient of 0.10, which is typical of that observed during the final load stage during each of the tests. However, unlike simple volatility measurement, the LFW-1 wear test should also be sensitive to antiwear additives. As a result, the LFW-1 wear test appears to work equally well with both the 25 commercially available and the six experimental lubricants.

IV. THE DIESEL ENGINE OIL SCUFF TEST (DEOST)

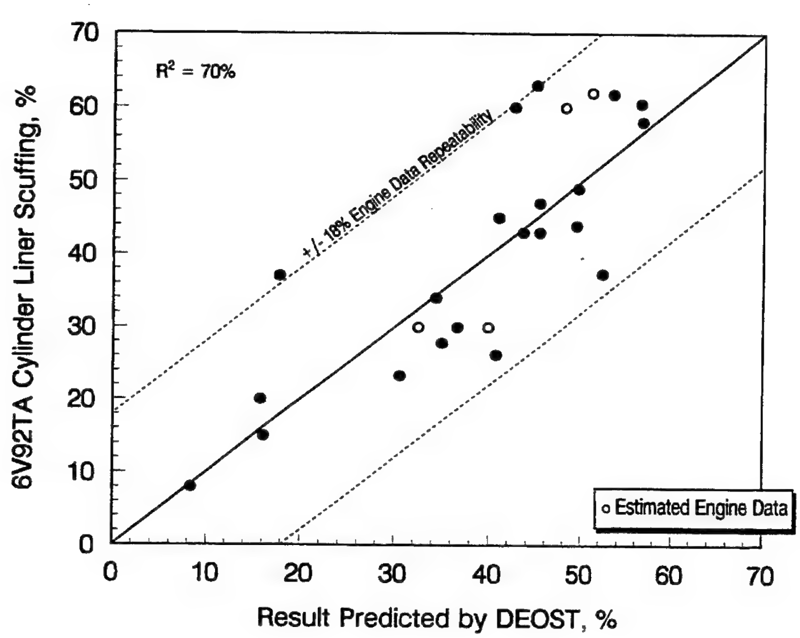
A. Combined Effects of Volatility and Wear

Clearly, the onset of scuffing in the full-scale engine is sensitive to the effects of both basestock volatility and antiwear additives. A multiple linear regression model was used to fit the LFW-1 wear test and the volatility results at 525°C to the 6V92TA engine data. The fitted model is described mathematically in Equation 2 and the DEOST results plotted in Fig. 7. An R^2 correlation of 70 percent was obtained for the complete matrix of 25 test lubricants. All of the data points fall on or within the 95 percent confidence limit for the full-scale engine test data.

$$X = 25.43(Y) + 0.606(Z) - 51.7 \quad \text{(Eq. 2)}$$

where

X = predicted 6V92TA engine test result by the DEOST (% scuffing)
Y = LFW-1 wear test result (mm)
Z = lubricant boiling point fraction at 525°C (wt%) per ASTM D 2887.



(Note: The hollow data points represent best estimates of the 6V92TA data using results obtained from the 6V53T.)

Figure 7. Correlation between the DEOST procedure and cylinder liner scuffing, as measured in the 6V92TA engine test

B. Application of the DEOST Procedure

Six advanced experimental lubricants intended for use in high-temperature military engines were evaluated using the DEOST procedure, with the results provided in TABLE 5. The experimental oils were synthetic-based and contained an unknown blend of synthetic components, solely intended for use in partially cooled compression ignition engines. These oils were formulated using high stability additive packages selected to minimize deposit formation. Indeed, Oils E-2 and E-3 did not contain zinc or phosphorous, which indicates that a conventional zinc dithiophosphate additive was not used. Nonetheless, each of the six experimental lubricants provided low wear in the LFW-1 apparatus and possessed excellent volatility characteristics. As a result, the DEOST procedure predicted that the oils would produce excellent results in the 6V92TA engine, with cylinder liner distress of less than 26 percent in each instance.

TABLE 5. Results Obtained for Advanced High-Temperature Lubricants

Oil Code	SAE Grade	LFW-1 Result, mm*	Four-Ball, mm	Kin. Visc., cSt 40°C	Kin. Visc., cSt 100°C	Visc. Index	B.P. Frac., wt% 371°C	B.P. Frac., wt% 525°C	DEOST Result, %
E-1	10W-30	1.00	0.36	56	10	167	4.5	87	26
E-2	30	1.08†	0.95	63.8	9.83	138	1.5	16	0
E-3	40	0.97	0.41	105.5	14.5	141	2.5	31	0
E-4	40	1.12†	0.39	65.9	13.6	214	4.6	78	24
E-5	40	0.78	0.39	101.5	14.9	153	2.5	92	23
E-6	40	1.07	0.41	101.8	12.9	122	2.5	52	7

* Average value calculated from duplicate tests.
† Initial test seized due to scuffing. Test was repeated without final load increment.

The good performance of these lubricants in the DEOST test is validated by practical experience. Indeed, no measurable scuffing was present with oils E-1, E-2, or E-3 when tested in the SCE 903 engine test, detailed in TABLE 2. Clearly, antiwear additives are not of primary importance in the 6V92TA engine test. In contrast, however, Oil E-2 produced severe wear when tested in the four-ball wear test apparatus.

A second matrix of test lubricants was assembled to better define the effects of lubricant additives on the LFW-1 results and by association, the 6V92TA. The test matrix is summarized in TABLE 6 and consists of fully formulated oils and the basestocks from which they are derived. The SAE oil grades include 10, 30, and 40W. In addition, an SAE 15W-40 was produced through inclusion of a viscosity index improver in the SAE 10W basestock. The fully formulated oils all contain the same conventional additive package, which includes approximately 2,000 ppm of both phosphorous and zinc. The kinematic viscosity, viscosity index, and boiling point distribution were also defined for each oil, with the results provided in TABLE 6.

TABLE 6. Effect of Kinematic Viscosity and Lubricity Additives

SAE Grade	Description	LFW-1	Kin. Visc., cSt 40°C	Kin. Visc., cSt 100°C	Visc. Index	B.P. Frac., wt% 371°C	B.P. Frac., wt% 525°C	DEOST Result, %
10W	Basestock	Seized	49	7.06	101	6	95	100
10W	Fully Formulated	Seized	45	6.6	97	13	97	100
10W	Basestock	Seized	49	7.06	101	6	95	100
15W-40	Basestock + VI Improver	Seized	89	12.23	131	7	95	100
15W-40	Fully Formulated	1.9	99	15.06	160	8	95	55
30	Basestock	Seized	75	9.21	97	3	90	100
30	Fully Formulated	1.57	86	10.8	110	5	90	43
40	Basestock	1.29	121	12.56	95	3	75	30
40	Fully Formulated	1.32	130	14.8	115	5	79	31

Note: The fully formulated oil consists of the basestock plus a conventional additive package with approximately 2,000 ppm of both phosphorous and zinc.

In general, the additive package for the single-grade oils produced a slight increase in viscosity and viscosity index while not significantly affecting the boiling point distribution. However, the additives did produce a measurable increase in scuffing resistance for medium-grade oils. Indeed, very severe scuffing and seizure occurred for each of the unadditized basestocks except the SAE 40 grade oil. Neither the antiwear additives nor the viscosity index improver were alone capable of preventing seizure in the SAE 10 grade basestock. Their combined effects did reduce wear, but not to an acceptable level. In contrast, the antiwear additives showed no measurable benefits

in the SAE 40 grade oil. This viscous oil has naturally high scuffing resistance and was predicted to pass the 6V92TA engine test even without use of antiwear additives.

V. DISCUSSION

In the present study, a laboratory-scale wear test procedure was combined with a measure of lubricant volatility to predict cylinder liner scuffing under severe operating conditions. The resulting Diesel Engine Oil Scuff Test (DEOST) produced an R^2 correlation of 70 percent with the 6V92TA engine test. This correlation is appreciably better than that obtained through consideration of either volatility or wear characteristics alone. Simple mathematical models predict similar contact temperatures of approximately 500°C in both the LFW-1 procedure and the piston-ring/cylinder-liner interface. As a result, the model is optimized when lubricant boiling point fraction is also measured at this temperature. In contrast, industry practice (ASTM D 4485) currently limits the lubricant fraction lost at 371°C to control engine oil volatility. However, in the present study, a relatively low R^2 correlation of 21 percent was observed between the 6V92TA and boiling point fraction at the specified temperature of 371°C, increasing to 60 percent at 525°C.

Previous workers have obtained good correlation between the LFW-1 apparatus and cylinder liner wear under less severe conditions than those produced by the 6V92TA.(35) However, this application required a revised procedure that produced high contact temperatures with little abrasive wear. To achieve this goal, polished specimens with a conforming contact geometry are used to allow a very high contact load, with a minimum of entrained lubricant. Relatively poor correlation was observed between the four-ball wear test and the full-scale engine. This low correlation is probably due to inappropriate contact conditions produced by the four-ball test. Indeed, mechanistic models typically predict a mean contact flash temperature of only 150 to 220°C in the four-ball apparatus, while consideration of chemical kinetics for reaction products observed around the contact predicts a somewhat higher temperature of 300 to 350°C.(36, 37) Both temperatures are significantly lower than the flash temperatures predicted to occur in either

the LFW-1 wear test or the full-scale engine. In addition, the relatively small counterformal four-ball contact minimizes the time available for volatization of the entrained lubricant.

Most current compression-ignition engines operate at a maximum cylinder liner temperature of approximately 180 to 200°C. Results from experimental work with insulated compression ignition engines have shown bulk metal temperatures between 150 to 600°C at the top ring reversal point (13) with appreciably higher contact flash temperatures. Lubricant development for future high-temperature engines has emphasized increased thermal stability and improved deposit ratings. However, lubricant deposits are commonly minimized in oils that contain few metal-based antiwear additives and so produced severe wear in simple boundary lubricated conditions such as the four-ball wear test. In addition, simple consideration of lubricant volatility did not always predict the performance of these oils. The results obtained using the DEOST procedure indicated that the advanced lubricants intended for use in partially cooled military engines should give excellent results in less demanding commercially available equipment.

The contact load at conclusion of the LFW-1 test is typically in the range of 180 to 350 N/mm^2, which is appreciably higher than the *average* value present beneath piston rings during normal operation. However, the contact load at the piston ring and liner interface may be significantly increased by thermoelastic distortion of the surfaces. Most importantly, the predicted flash temperature is very similar in each instance due to the higher sliding speed of the piston ring and liner contact. Clearly, the DEOST procedure is most sensitive to basestock characteristics, particularly volatility. Petroleum mineral oils are the most commonly used basestock in commercial lubricants due to their relatively low cost. Moreover, their viscosity characteristics are often adjusted using viscosity index improvers, which were shown to be largely ineffective in the present application. In contrast, ester-based lubricants typically have good oxidative stability and additive response and so are ideally suited to high-temperature applications. Indeed, advanced synthetic lubricants that contain few antiwear additives produced excellent results in both the DEOST and full-scale engine tests.

The full-scale engine test remains the more effective method of evaluating lubricant performance, as all engine operating variables may never be duplicated in the laboratory. However, the

DEOST procedure provides a cost effective and repeatable mechanism of screening lubricant additive and basestock formulations prior to the performance of full-scale engine tests. The present limited data set would indicate that the DEOST procedure is normally capable of correctly predicting cylinder liner scuffing to within the test repeatability of the 6V92TA engine.

VI. CONCLUSIONS

The following conclusions may be drawn from the present study in relation to cylinder liner scuffing in the 6V92TA engine test:

a) A laboratory procedure (the DEOST procedure) was developed that can predict cylinder liner scuffing in the 6V92TA engine test with an R^2 correlation of 70 percent. The DEOST procedure is sensitive to both oil volatility and boundary lubricating characteristics.

b) Cylinder liner scuffing is primarily sensitive to basestock characteristics, particularly volatility.

c) Antiwear additives have a secondary but measurable effect on scuffing, particularly for oils of borderline quality. Antiwear additives had no measurable effect on less volatile oils. The effects of antiwear additives are not apparent from simple measurement of lubricant volatility.

d) An R^2 correlation of 60 percent with the 6V92TA engine test was obtained with boiling point fraction measured at approximately 525°C. However, the correlation was reduced to 40 percent if experimental lubricants with an unconventional boiling point distribution were considered.

e) Poor correlation was observed between cylinder liner scuffing and boiling point fraction measured at 371°C, as specified in ASTM D 4485.

f) In the present application, lubricant viscosity affects scuffing resistance, both through hydrodynamic lift and its indirect relationship to volatility.

g) Because of Items b) and f), viscosity index improvers show only marginal benefits, and multigrade oils typically produce poor results.

h) Results obtained using the DEOST procedure indicated that the advanced lubricants intended for use in partially cooled military engines should give excellent results in less demanding commercially available equipment.

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APPENDIX

LFW-1 Wear Test as Used in the DEOST Procedure

Test Method to Predict Cylinder Liner Scuffing Performance of Lubricants in a Turbocharged Two-Cycle Diesel Engine (6V92TA) Test

Draft 1.1
22 August 1996

1. Scope

1.1 This test method predicts the level of cylinder liner distress measured in a turbocharged two-cycle diesel engine operated with the test oil.

1.2 The method employs a block-on-ring friction and wear testing machine.

1.3 The method is applicable to lubricating oils intended for use in compression ignition engines and is primarily sensitive to the effects of volatility and surface active antiwear additives.

1.4 The values stated are in SI (metric) units.

1.5 *This standard may involve hazardous materials, operations, and equipment. This standard does not purport to address all of the safety problems associated with its use. It is the responsibility of the user of this standard to establish appropriate safety and health practices and determine the applicability of regulatory limitations prior to use.*

2. Referenced Documents

2.1 *ASTM Standards:*

D 2714 Calibration and Operation of the Falex Block-on-Ring Friction and Wear Testing Machine[1]

D XXXX-XX Standard Test Method for Evaluating Lubricity of Diesel Fuels by the Scuffing Load Ball-on-Cylinder Lubricity Evaluator (SLBOCLE)[2]

3. Terminology

3.1 *Descriptions of Terms Specific to This Standard:*

3.1.1 *applied load, n*—the weight in grams added to the load arm of the block-on-ring wear test machine.

3.1.2 *contact load, n*—the force in grams with which the block contacts the test ring.

3.1.2.1 *Discussion*—For the Falex Block-on-Ring Wear Test Machine load system, the contact load is thirty times the applied load.

3.1.3 *scuffing, n*—in lubrication, damage caused by instantaneous localized welding between surfaces in relative motion which does not result in immobilization of the parts.

4. Summary of Test Method

4.1 A 40-ml test sample of oil is placed in the test reservoir of a block-on-ring apparatus and adjusted to the pretest temperature of 70°C.

4.2 A polished steel test ring is rotated against a stationary steel test block, the specimen assembly being partially immersed in the lubricant sample. The velocity of the test ring is 66 ± 0.55 m/min, which is equivalent to a spindle speed of 600 ± 5 r/min.

4.3 The applied load is increased periodically to account for the wear process. The specimens are ultimately subjected to a contact load of 150 kg applied by 5 kg of dead weight on the 30:1 ratio lever system. Total test duration is 12,000 cycles, which requires 20 minutes.

4.4 The average width of the wear scar on the stationary block at the end of the test is measured.

4.5 The complete test is repeated three times. An average wear scar width is used as a measure of lubricant quality.

5. Significance and Use

5.1 This test method may be used to predict the degree of protection provided by the test fluid against scuffing damage to the cylinder liner in diesel engines.

5.2 Correlation has been indicated with liner distress in the Detroit Diesel 6V92TA industrial engine, tested in accordance with ASTM D 5862, as indicated in the Appendix.

5.3 The user of this test method should determine to his or her own satisfaction whether results of this test method correlate with field performance or other bench test machines. If the test conditions are changed, wear values can change and relative ratings of fluids can be different.

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[1] *Annual Book of ASTM Standards*, Vol. 05.03.

[2] Available from Southwest Research Institute, 6220 Culebra Road, San Antonio, TX 78228, (210) 522-3367. Procedure currently being balloted.

6. Apparatus

6.1 *Falex Block-on-Ring Test Machine*[3]

6.1.1 The Falex Block-on-Ring Test Machine is similar to that specified in Test Method D 2714. Complete operating conditions are listed in Table 1.

NOTE 1: Consult the instruction manual for each machine to determine respective capabilities and limitations.

TABLE 1 Operating Conditions

Fluid Volume	40 ± 1 ml
Pretest Fluid Temp.	70 ± 2°C
Cylinder Rotational Speed	600 ± 10 r/min
Test Duration	
Distance	12,000 rev
Time	20 min
Applied Load	
0 to 6,000 rev	3,000 g
6,000 to 10,000 rev	4,000 g
10,000 to 12,000 rev	5,000 g

6.1.2 A stationary rectangular test block is pressed with a predetermined load against a rotating ring (Fig. 1).

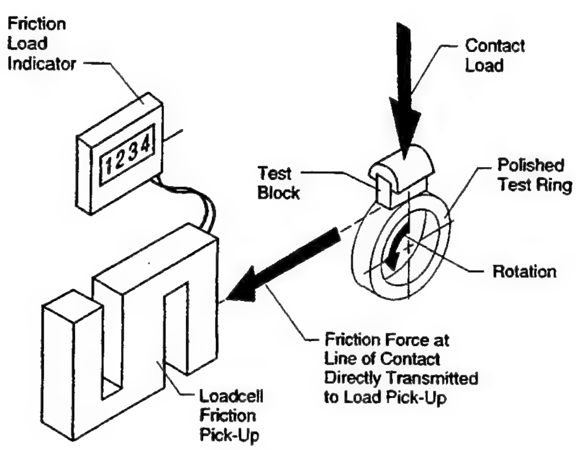

Fig. 1 Functional Diagram of the Falex Block-on-Ring Test Machine

6.1.3 A counter records the number of revolutions of the test specimen.

6.1.4 The test shaft of the machine is supported by two roller bearings, and the mandrel end of the shaft protrudes through the front panel of the machine where the test specimens are mounted.

6.1.5 The test block is held stationary against the revolving ring and is restrained from horizontal movement. The design of this block holder allows the test block to align itself automatically in a manner prescribed by ASTM specifications for compression-loaded specimens. This maintains uniform loading throughout the area of contact between the specimens, regardless of the force existing between them.

6.1.6 The normal force between the test specimens is produced by hanging dead weights on the lower end of a lever system that is designed in such a way as to allow the full value of the friction force to be transmitted to the frictional load pick-up device.

6.1.7 An electronic six-digit cycle counter is mounted on the front of the digital instrumentation unit.

6.2 *Measuring Magnifier Glass,* with SI calibration so that the scar width can be measured with a precision of 0.01 mm.

7. Reagents and Materials

7.1 *Test Rings,* Falex Type S-10,[4] SAE 4620 carburized steel, having a hardness of 58 to 63 HRC. The test ring has a width of 8.15 mm, a diameter of 35 mm, and a maximum radial run-out of 0.013 mm.

7.1.1 The test rings are given a polished surface finish, similar to that described in ASTM D XXXX-XX.[4]

7.2 *Test Blocks,* Falex Type H60, SAE 01 tool steel having two ground test surfaces of 0.10 to 0.20 µm rms. The test block has a total test surface width of 6.35 mm and a length of 15.76 mm. The test block has a hardness of 58 to 63 HRC.

7.2.1 A 2-mm wide groove is formed at the center of the block using an electronic discharge machine, as shown in Fig. 2.[4]

7.3 *Solvents,* safe, nonfilming, nonchlorinated.

NOTE 2: Each user should select a solvent that can meet the applicable safety standards and still thoroughly clean the parts.

7.4 *Calibration Fluid,* consisting of white mineral oil conforming to U.S. Pharmacopeia XVII, p. 399, and with a viscosity of 63 to 65 cSt at 37.8°C.

[3] The block-on-ring test machine is manufactured by Falex Corporation, 2055 Comprehensive Dr., Aurora, IL 60505.

[4] Available from Southwest Research Institute, 6220 Culebra Road, San Antonio, TX 78228, (210) 522-3367. Procedure currently being balloted.

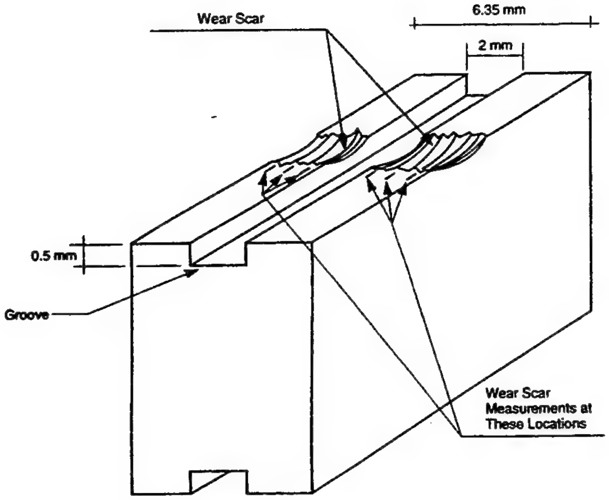

Fig. 2 Test Block

8. Preparation of Apparatus

8.1 Before each test, thoroughly clean the specimen holder and chamber as well as the tapered section, threaded section, lock nut, lock washer, a new test ring, and block using solvents selected in Section 7.3.

8.2 Place the block holder on the block. Carefully place block and holder in upper specimen holder in test chamber.

8.3 *Test Ring*

8.3.1 Mount the test ring on the test shaft, taking care not to touch the test area. Tighten the test ring on the shaft with 27 N/m torque.

8.3.2 Apply a small amount of test lubricant to the outer surface of the test ring.

8.3.3 Gently apply 500-g load to the bale rod.

8.3.4 Operate motor at 30 r/min.

8.3.5 Confirm concentricity of test ring is such that less than 0.5 mm of movement of the bale rod is measured using a dial indicator.

8.3.6 If excessive movement is observed, rotate test ring on shaft by approximately 45°.

8.3.7 Repeat from Section 8.3.6 until total run-out is minimized. If bale movement cannot be reduced to less than 0.5 mm, the test ring should be replaced and the procedure repeated from Section 8.3.1.

8.4 Remove 500-g applied load and insert locking pin in bale rod.

8.5 Mount the reservoir and heater block in position.

NOTE 3: If an alternate or nonstandard reservoir is used, the test fluid should be approximately halfway on the spindle. This volume must be measured and the same amount used for each test.

8.6 Transfer 40 cu. cm of fluid to the reservoir.

8.7 Set the temperature control of the oil reservoir to 70°C.

9. Calibration

9.1 A machine shall be judged to be in acceptable condition when the average wear measurement obtained with the calibration fluid falls within a range yet to be defined.

10. Procedure

10.1 Start the machine and bring the speed to 600 r/min.

10.2 Gently place a 3-kg load on the bale rod.

10.3 When the fluid reaches 70°C, remove the locking pin from the bale rod and gently lower the weights, being very careful to avoid shock-loading.

NOTE 4: A considerable increase in temperature may occur during testing due to dissipation of frictional energy.

10.4 Increase the applied load to 4 kg at 6,000 revolutions and check the speed.

10.5 Increase the applied load to 5 kg at 10,000 revolutions and check the speed.

10.5.1 If severe scuffing is observed at 5-kg applied load, the test may be repeated without the final load increment.

NOTE 5: Severe scuffing is indicated by tangential friction force exceeding 30 kg.

10.6 Stop the machine at 12,000 revolutions and *immediately* record end-of-test oil temperature.

10.7 Remove the test block and test ring and clean them thoroughly using solvents selected in Section 7.3.

10.8 *Measurement of Wear Scar*

10.8.1 Measure the width of the wear scar at three points on the test surface on either side of the 2-mm wide groove on the test block.

10.8.2 The measurements on each side of the groove should be taken at the center and approximately 0.1 mm away from each edge, as shown in Fig. 2.

10.8.3 Record the average of the six measurements.

10.9 Three repeat tests are required to establish a satisfactory average.

10.9.1 The repeat tests should be performed with clean new metallic test specimens and test fluid. However, the reservoir need not be cleaned.

11. Calculation and Report

11.1 Report the following information:

11.1.1 Average wear scar width from three complete repeat tests, as determined in Section 10.9.

11.1.2 Applied load used at end of test, normally 5 kg unless scuffing is observed, as described in Section 10.5.

11.1.3 End-of-test oil temperature.

11.1.4 Description of the test oil.

12. Precision and Bias

12.1 *Precision*—The following criteria should be used for judging the acceptability of results (95% confidence):

12.1.1 *Repeatability*—The difference between successive test results obtained by the same operator with the same apparatus under constant operating conditions on identical test material would, in the long run, in the normal and correct operation of the test method, exceed the following value in only one case in twenty:

Repeatability = to be determined

12.1.2 *Reproducibility*—The difference between two, single and independent results obtained by different operators working in different laboratories on identical test material would, in the long run, in the normal and correct operation of the test method, exceed the following value in only one case in twenty:

Reproducibility = to be determined

12.2 *Bias*—Since there is no accepted reference material suitable for determining the bias for the procedure, bias has not been determined.

APPENDIX

(Nonmandatory Information)

X1. CORRELATION OF TEST METHOD WITH CYLINDER LINER DISTRESS MEASURED DURING ENGINE TESTS

X1.1 The test method may be used to predict the degree of protection provided by the test fluid against scuffing damage to the cylinder liner in diesel engines.

X1.2 Correlation has been indicated to exist with liner distress in the Detroit Diesel 6V92TA industrial engine, tested in accordance with ASTM D 5862, as indicated in Fig. X1.1.

NOTE X1: The data presented for the block-on-ring friction and wear test machine are the result of single tests, not three repeat tests as described in Section 10.9.

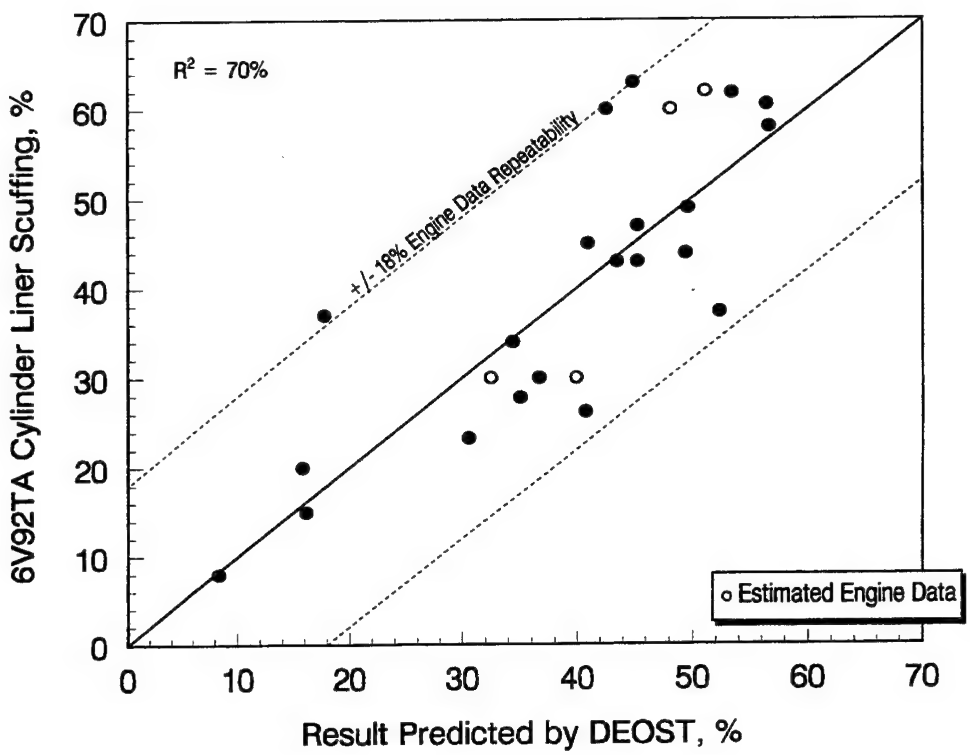

Fig. X1.1 Correlation Achieved Between Test Method and Cylinder Liner Distress Observed in the 6V92TA Diesel Engine

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